Investigation of Vehicle Body Stiffness Modification on Vehicle Ride Comfort and Handling Using a Low-Order Vehicle Model

UNDERGRADUATE HONORS THESIS

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Abstract

Vehicle body stiffness is an important design consideration with respect to vehicle ride comfort and handling performance; however, the interaction between the stiffness of the vehicle body and the suspension layout is not well understood. This complex interaction is often evaluated using complex finite element models or experimental techniques which are resource-intensive, especially early in the development process. Therefore, the objective of this research is to develop a simplified vehicle model with a vehicle body stiffness representation to better understand its effect on ride comfort and handling performance. First, a low-order multibody dynamics vehicle model is created with vehicle suspension kinematics, vehicle body stiffness, realistic spring rates, and shock absorber performance curves from published or modally justified parameters. Next, the functionality of the model is verified by comparison to published static and dynamic vehicle test data. Virtual test profiles and assessment criteria are then defined for the models to simulate and assess vehicle ride and handling phenomena. Finally, the influence of vehicle body stiffness modifications is quantified using the model by systematically varying the vehicle body stiffness. Results of this study identified different vehicle performance regime changes related to modifications of vehicle body stiffness. The effects of these changes are compared to changes due to realistic variations in tire and shock absorber properties to quantify their significance. For the vehicle considered, improvements to both ride and handling could be achieved through decreasing vehicle body stiffness by upwards of 50%; however, in comparison to realistic

variations in tire and shock absorber parameters, the effects of modifications to the body stiffness are minimal. Although changes in vehicle body stiffness are found to be insensitive as part of this study, the tractable modeling approach from this research could be used in low-order vehicle design tools to quickly assess the influence of vehicle body stiffness on the ride comfort and handling performance of future vehicle designs.

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1. Introduction

The automotive industry is increasingly relying on virtual tools during the vehicle development stage in an effort to reduce development cycle times and the number of costly physical prototypes. Early in the development stage, vehicle performance targets are set. A variety of models, varying in complexity, are used to set and physically realize these targets. Physical prototype vehicles are built to further tune the product to meet the targets. In parallel to this activity, additional modeling and test iterations and refinements are conducted to address unanticipated issues, such as noise or vibration from system interactions or production variation.

The effect of suspension layout on vehicle ride comfort and handling performance is well known [1]. However, the interaction between the stiffness of the vehicle body and the suspension layout is not well understood. This complex interaction is often evaluated using complex finite element models [2] or experimental techniques [3]. Due to resource-intensive nature of these studies, the results are often limited to a select few cases and are not ideal for early stage development, where vehicle specifications are set.

Some reduced-order lumped parameter models, described in [4] and [5], are commonly used in the automotive industry to evaluate chassis and suspension targets with respect to vehicle ride comfort and handling. Often these two phenomena are evaluated using separate models. These models often ignore or lump the contribution of vehicle body stiffness into suspension parameters. The models often do not include enough fidelity to evaluate interactions among suspension layout parameters, such as spring rates, damper profiles, suspension kinematic relationships, and vehicle body stiffness. The inclusion of this vehicle body stiffness has become increasingly important due to the automotive industry trend of light-weighting of vehicle structures [2].

In order to capture the effect of the suspension layout, a multibody dynamics (MBD) approach is often used [1]. These higher fidelity models require vehicle suspension layout information in addition the inertial properties of the bodies and the elastic properties of the joints within the suspension. These are the models that are often used to determine how to achieve desired vehicle performance through modifications of the suspension system. Body stiffness can be incorporated into these models through the addition of intermediate bodies and elastic joints or through modal substructures generated through finite element model reduction [6]. However, selection of proper parameters and model partitioning as well as integration into the multibody environment is resource-intensive. Therefore, new reduced-order models will be needed to study this phenomenon early in the development process.

2. Problem Formulation and Methodology

2.1. Problem Formulation

The goal of this research is to develop a simplified vehicle model with a vehicle body stiffness representation to better understand its effect on ride comfort and handling performance. Accordingly, the steps to achieve the research goal are to:

(i) create a low-order vehicle MBD model with vehicle suspension kinematics, vehicle body stiffness, realistic spring rates, and shock absorber performance curves from published or modally justified parameters;

(ii) verify the functionality of the model by comparison to published static and dynamic vehicle test data;

(iii) define virtual test profiles and assessment criteria for the models to simulate and assess vehicle ride and handling phenomena;

(iv) and quantify the influence of vehicle body stiffness modifications by systematically varying the vehicle body stiffness.

2.2. Development of a Low-order Vehicle Model

First, a low-order multibody dynamics model is developed in MATLAB SimMechanics [7]. The vehicle coordinates follow the following convention: positive X-axis points to rear of vehicle; positive Y-axis points from driver side to passenger side;

and Z-axis points from bottom to top of vehicle. The origin for this vehicle coordinate system is assumed to be at the center of the front axle. Bodies and joints are defined according to this coordinate system. The model is composed of a simplified swing-arm suspension representation. The swing arm connection locations on the vehicle body are located at the instant centers related to the wheel motions as constrained to the vehicle body by the suspension kinematics in Y-Z plane, illustrated in Figure 1.

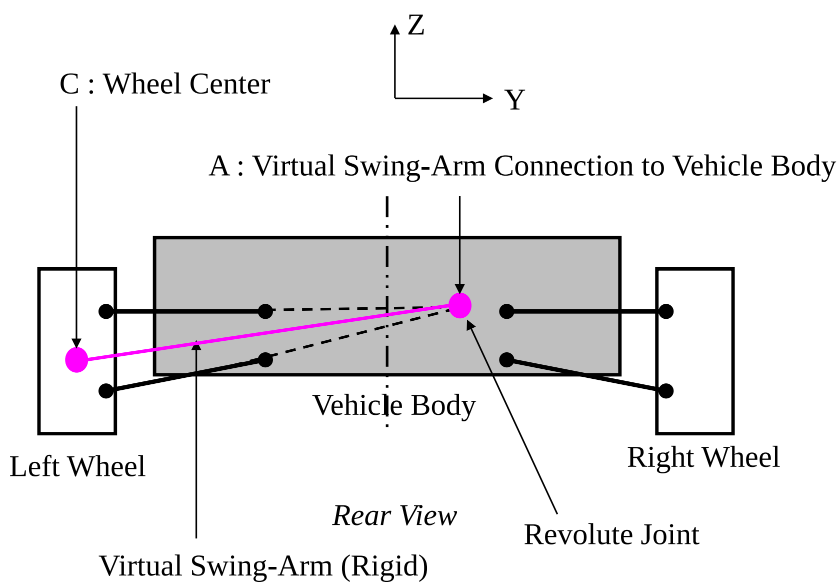

Figure 1. Example of a Virtual Swing Arm Suspension Representation

The suspension parameters, such as wheel inertia, spring rates and damper performance curves (force vs. velocity), and body connection locations, are obtained from an existing MBD vehicle model detailed by Blundell and Harty [1]. The vehicle is an automotive sedan with a double-wishbone type front suspension and a trailing arm / strut type independent rear suspension. This model [1] considers the vehicle body to be a singular rigid body and connected to the suspension bodies using constrained joints.

For this type of vehicle, the first flexural mode of the vehicle body is assumed to be a torsional mode of the vehicle (effectively twisting along the X-axis of the vehicle body). Therefore, in order to represent the vehicle body stiffness in the model, the vehicle body is discretized into three bodies, each with its own center of mass and inertia, and connected by a revolute joint (along the roll axis of the vehicle) with associated torsional stiffness and damping properties. The simplified tire contact patch of each wheel is connected to an actuator pad at the effective roll radius of the tire through a Cartesian joint with stiffness and damping properties in the X, Y, and Z directions. The actuators can be independently displaced with either a lateral or vertical motion required to simulate either a vertical ride input or a lateral handling input. Dynamic measurements of the vehicle body motion will be made at the center of mass of the middle body, quantifying nominal effects observed by the driver and passengers. This model concept is illustrated in Figure 2.

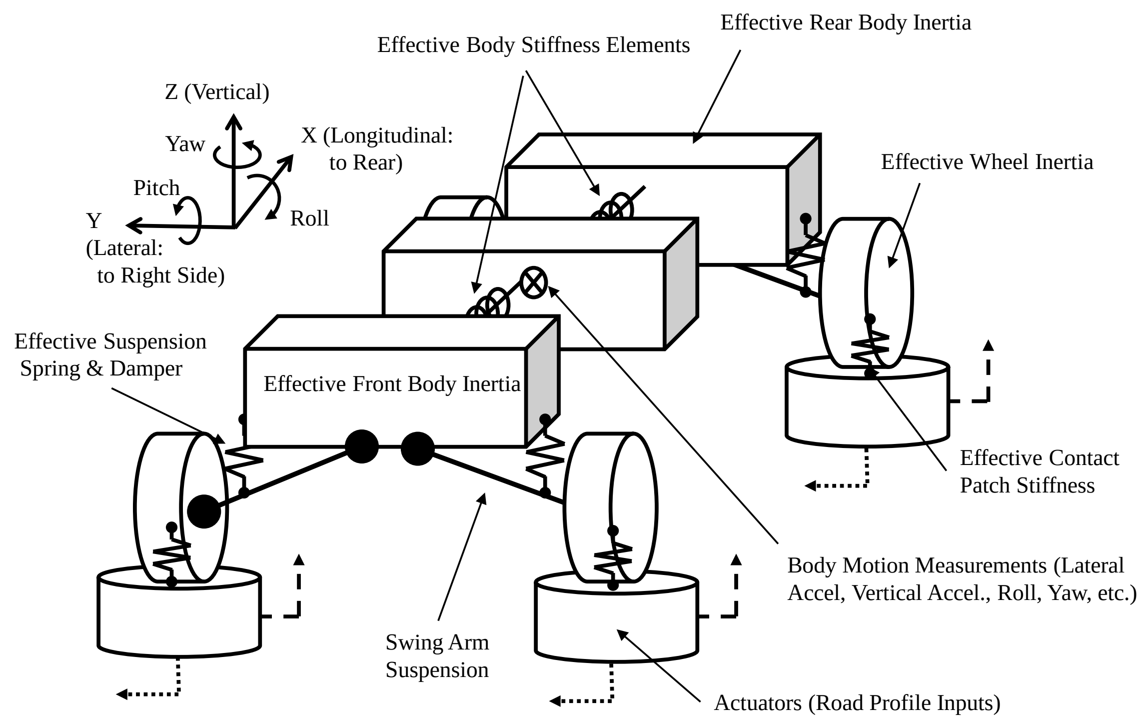

Figure 2. Low-order Multibody Dynamics Model Concept with Simple Swing Arm Suspension Layout and Vehicle Body Stiffness

As stated before, to incorporate the vehicle body stiffness, the vehicle is partitioned into three discrete bodies connected revolute joints free to rotate about the X-axis of the vehicle. Each of these three discrete mass elements are assumed to have equivalent mass and mass moments of inertia. The center of mass of the front body CM_1 is lumped at the center of the front axle at the same height as the vehicle center of mass. The center of mass of the rear body CM_3 is lumped at the center of the rear axle at the same height of the vehicle center of mass. If assembled by rigid connections, the total mass of the vehicle (m_t) and mass moments of inertia (I_{xxt}, I_{yyt}, and $\mathrm{I}_{\mathrm{zzt}}$ about the X-, Y-, and Z-axes, respectively) are equivalent to the values reported in literature [1] for the full vehicle body at its center of mass (CM_2), also denoted as the location of the center of mass for the middle body.

Accordingly, the mass parameters for the different discretized bodies are related to the total vehicle body mass by the following relationship:

$$m_t = m_1 + m_2 + m_3 \tag{1}$$

where m_1 is the mass of the front section of the vehicle body lumped at point CM_1 located at (x_1, y_1, z_1), m_2 is the mass of the middle section of the vehicle body lumped at point CM_2 located at (x_2, y_2, z_2), and m_3 is the mass of the rear section of the vehicle body lumped at point CM_3 at (x_3, y_3, z_3). The mass moments of inertia (about the X-axis located along the center of mass of the vehicle) of each vehicle body section are related to the corresponding total mass moment of inertia is given by:

$$I_{xxt} = I_{xx1} + I_{xx2} + I_{xx3} \tag{2}$$

where I_{xx1} is the mass of the front section of the vehicle body lumped at point CM_1, I_{xx2} is the mass of the middle section of the vehicle body lumped at point CM_2 , and I_{xx3} is the mass of the rear section of the vehicle body lumped at point CM_3. Likewise, the mass moments of inertia, about the Y-axis located along the center of mass of the vehicle and about the Z-axis located along the center of mass of the vehicle, for each section are related to the corresponding total mass moments of inertia by the following relationships:

$$I_{yyt} = I_{yy1}\left(x_1 - x_2\right)^2 + I_{yy2} + I_{yy3}\left(x_3 - x_2\right)^2 \tag{3}$$

and

$$I_{zzt} = I_{zz1}\left(x_1 - x_2\right)^2 + I_{zz2} + I_{zz3}\left(x_3 - x_2\right)^2 \ , \tag{4}$$

respectively.

The vehicle body torsional stiffness parameters, $k_{tD,FR}$ connecting the front and middle bodies and $k_{tD,RR}$ connecting the middle and rear bodies, are determined by matching the frequency range (30-40 Hz) associated with the first torsional flexural mode (without suspension) of a similar vehicle body from Rashid et al. [8]. For the nominal condition of this vehicle, the values of $k_{tD,FR}$ and $k_{tD,RR}$ are assumed to be equivalent; however, they will be changed independently as part of design studies presented later in the thesis. The vehicle body torsional viscous damping parameters, $c_{tD,FR}$ connecting the front and middle bodies and $c_{tD,RR,}$ connecting the middle and rear bodies, are assumed to be proportional to associated stiffness parameters.

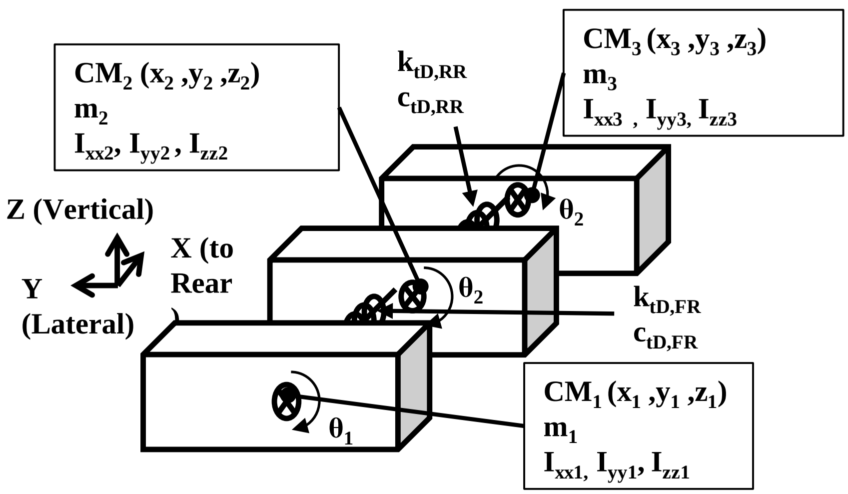

Figure 3. Discretized Vehicle Body Representation

In order to calculate the value of vehicle body stiffness, natural frequencies of the discretized vehicle body with assumed stiffness parameters are first calculated. First, each of the discretized vehicle bodies are free to rotate about the X-axis at the center of mass of the vehicle, as shown in Figure 3. The displacement vector is given as

$\boldsymbol{\theta} = \{\theta_1 \quad \theta_2 \quad \theta_3\}^T$, composed of the angular motion of each degree of freedom (DOF). The associated inertia matrix is given as

$$\mathbf{I} = \begin{bmatrix} I_{xx1} & 0 & 0 \\ 0 & I_{xx2} & 0 \\ 0 & 0 & I_{xx3} \end{bmatrix}, \tag{5}$$

and the associated torsional stiffness matrix is given by

$$\mathbf{K_t} = \begin{bmatrix} k_{tD,FR} & -k_{tD,FR} & 0 \\ -k_{tD,FR} & k_{tD,FR} + k_{tD,RR} & -k_{tD,RR} \\ 0 & -k_{tD,RR} & k_{tD,RR} \end{bmatrix}. \tag{6}$$

For this proportionally damped system, to determine the natural frequencies, the homogenous, undamped form of the system equation is needed. The homogenous form is given by

$$\left(-\omega^2\mathbf{I} + \mathbf{K_t}\right)\{\boldsymbol{\theta}\}e^{j\omega t} = 0\,. \tag{7}$$

Solving the corresponding eigenvalue problem given by

$$\omega_i^2\mathbf{I}\{\boldsymbol{\theta}\}_i = \mathbf{K_t}\{\boldsymbol{\theta}\}_i \qquad\qquad i = 1, 2, 3 \tag{8}$$

gives ω_i as the i^{th} natural frequency of the system (in rad/s) and $\{\boldsymbol{\theta}\}_i$ as the corresponding mode shape vector to the natural frequency.

Additional details of the model parameters are shown in Figure 4. The values, descriptions, and sources of parameters are given in Table 1.

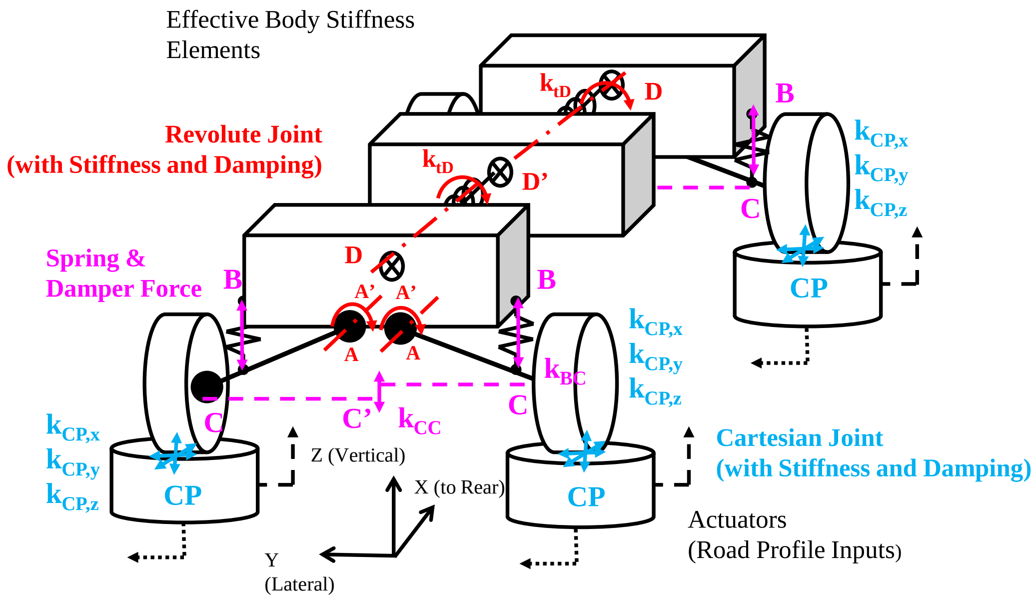

Figure 4. Model Layout with Parameter Conventions

Table 1. Descriptions, Values, and Sources of Model Parameters

* Calculated from MBD model in literature [1]
** Assumed or defined for this model
*** Calculated to match modal data in literature [8]

x, y, z : Defined in vehicle coordinates

	Type	Description	Parameter	Units	Front (FR) Left Hand (LH)	Front (FR) Right Hand (RH)	Rear (RR) Left Hand (LH)	Rear (RR) Right Hand (RH)	Source
Connections	Hard Points	Virtual Swing Arm Connection to Vehicle Body	A : (x_A, y_A, z_A)	mm	[0, 1364, 192.7]	[0, -1364, 192.7]	[2766, 577, 40.3]	[2766, -577, 40.3]	*
		Spring / Damper on Vehicle Body over Wheel Center	B : (x_B, y_B, z_B)	mm	[0, -744, 0]	[0, 744, 0]	[2766, -744, 0]	[2766, 744, 0]	*
		Spring / Damper on Suspension at Wheel Center	C : (x_C, y_C, z_C)	mm	[0, -744, 0]	[0, 744, 0]	[2766, -744, 0]	[2766, 744, 0]	*
		Contact Patch	CP : (x_{CP}, y_{CP}, z_{CP})	mm	[0, -744, -313]	[0, 744, -313]	[2766, -744, -313]	[2766, 744, -313]	*
		Center of Axle on Vehicle Body	D: (x_D, y_D, z_D)	mm	[0, 0, 286]		[2766, 0, 286]		**
	Stiffness	Suspension Stiffness at Wheel Center	k_{BC}	N/mm	$32/(1.43)^2$		$61/(1.91)^2$		*
		Contact Patch Stiffness	k_{CPx}, k_{CPy}, k_{CPz}	N/mm	[273, 113, 150]				**
		Vehicle Body Stiffness	k_{tD}	Nmm/rad	5000		5000		***
		Anti-Roll Bar Stiffness Between Wheel Centers	k_{CC}	N/mm	5.51		3.36		*
	Damping	Suspension Damping (Force vs. Velocity) Curve at Wheel Center	f_{BC} v_{BC}	N mm/s	$[-2400\ -1200\ 0\ 200\ 1100]/(1.43)^2$ [-1000 -150 0 100 1000]		$[-1100\ -488\ 0\ 125\ 350]/(1.30)^2$ [-1000 -150 0 100 1000]		*
		Contact Patch Damping	c_{CPx}, c_{CPy}, c_{CPz}	Ns/mm	[0.27, 0.11, 0.15]				**
		Vehicle Body Damping	c_{tD}	Nmms/rad	5		5		**
Bodies	Center of Mass	Center of Mass of Front Vehicle Body	CM_1	mm	[0, 0, 286]				**
		Center of Mass of Middle Vehicle Body	CM_2	mm	[1184, 0, 286]				**
		Center of Mass of Rear Vehicle Body	CM_3	mm	[2766, 0, 286]				**
	Mass	Mass of Front Vehicle Body	m_1	kg	475				**
		Mass of Middle Vehicle Body	m_2	kg	475				**
		Mass of Center Vehicle Body	m_3	kg	475				**
		Mass of Wheel & Suspension	m_w	kg	48		45		*
	Mass Moment of Inertia	Mass Moment of Inertia of Front Vehicle Body	I_{xx1}, I_{yy1} , I_{zz1}	kgm^2	[126, 132, 132]				***
		Mass Moment of Inertia of Middle Vehicle Body	I_{xx2}, I_{yy2} , I_{zz2}	kgm^2	[126, 132, 132]				***
		Mass Moment of Inertia of Rear Vehicle Body	I_{xx3}, I_{yy3} , I_{zz3}	kgm^2	[126, 132, 132]				***
		Mass Moment of Inertia of Wheel & Suspension	I_{xxw}, I_{yyw} , I_{zzw}	kgm^2	[1.34, 2.35, 1.34]		[1.26, 2.20, 1.26]		**

As mentioned before, the model is implemented in MATLAB SimMechanics [7]. Each of the vehicle bodies are created with respect to the global reference frame, allowing the hard points to be changed, rather than through tedious modifications of relative coordinate systems. A gravitational field is applied to the vehicle. The natural lengths of the spring elements, connecting points B and C on Figure 4 or at point CP connecting the actuator to the wheel body are defined to provide sufficient preload to maintain the defined hard point locations defined in Table 1.

The springs and damping force elements are implemented in the model at the wheel center, requiring a scaling of the stiffness or damping values used in the model by associated lever ratios, calculated from the suspension kinematics. The damper is implemented as a force element through the joint connecting points B and C. The damper performance profile is defined as a 5 point look-up table (with linear interpolation and extrapolation) relating reaction force to joint velocity. The anti-roll stiffness of the vehicle is implemented as a joint stiffness connecting the left and right wheel centers of either the front or rear of the vehicle. As both wheels travel in the same vertical direction with respect to the vehicle body, this joint stiffness does not affect the ride stiffness. As both wheels travel in opposite vertical directions with respect to the vehicle body, this joint stiffness reacts the opposite wheel.

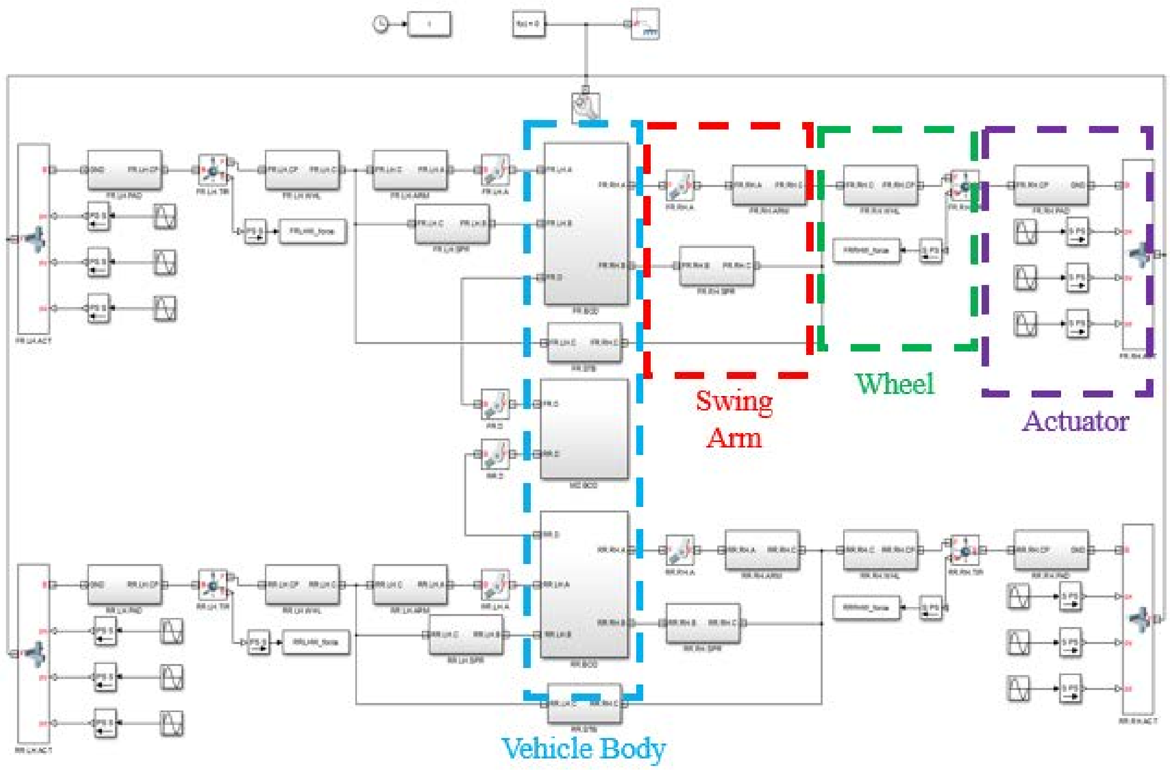

Figure 5. Full Vehicle Model MATLAB SimMechanics Schematic

The final vehicle model representation of the implemented model can be seen in Figure 5. Here, sub-models are grouped and defined for the vehicle body, swing arm, wheel, and actuator. The model is parametrized, including actuator input constraints and motion profiles, to facilitate design studies in later sections. A graphical representation of the model is also generated in MATLAB SimMechanics to observe the motions of the vehicle, shown in Figure 6. The motions of the center of mass of the middle body with respect to the ground reference frame are output to the MATLAB workspace for analysis in later sections.

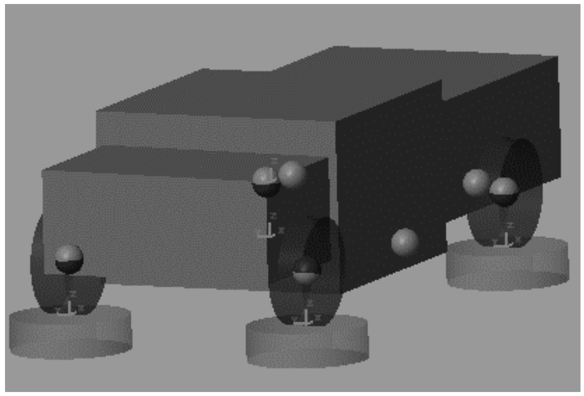

Figure 6. Graphical Representation in Full Vehicle Model in MATLAB SimMechanics

2.3. Verification of the Model Functionality

The functionality of the vehicle model is verified by evaluating vehicle model's static and dynamic response to the designed input. In order to evaluate vehicle static response to the input, the vehicle body elements are fixed to the global reference frame (ground). Next, a slow (0.1 rad/s) in-phase sinusoidal vertically motion is applied to the wheels through all four actuators. The actuators are free to move in the XY plane, simulating a typical kinematics and compliance vehicle test. The displacement of the front wheel is recorded as well as the reaction force through the tire contact patch. These values are then compared with the wheel rate data in the literature [1] for this vehicle. Figure 7 shows the result of vehicle model's static response using the MATLAB SimMechanics model. The slope of this curve, 15.6 N/mm, is consistent with the effective front suspension stiffness (k_{BC}) in the model from Table 1 as well as the value from literature [1].

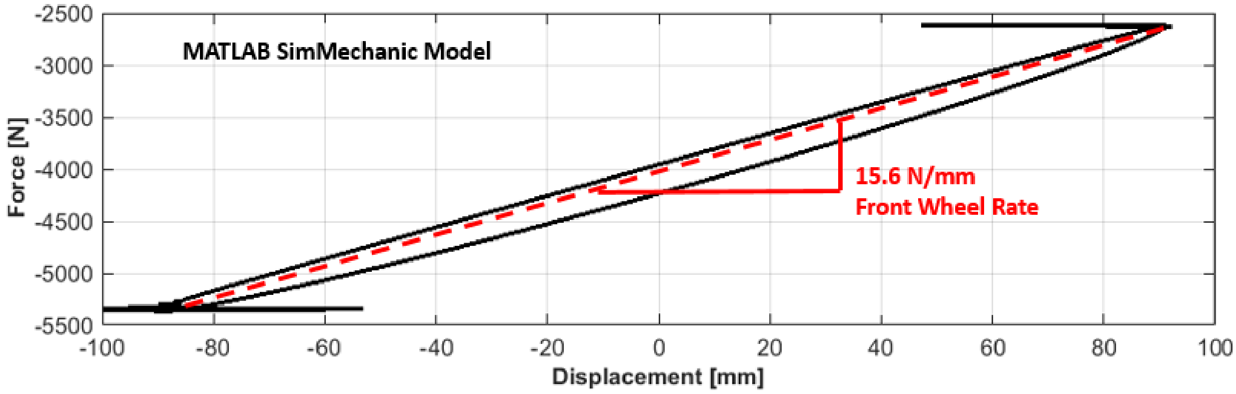

Figure 7. Vertical Force vs. Vertical Displacement with respect to Vehicle Body for Front Right Wheel

In order to evaluate vehicle model's dynamic response to the input with the vehicle body unconstrained, a stepped sine (0.5 Hz resolution) frequency sweep from 0.5 to 15 Hz of a constant 10 mm vertical displacement input is applied on all four actuators. This is commonly considered a heave input from a 4-post shaker vehicle test. The displacement amplitudes of wheels and vehicle center of mass are recorded at each discrete excitation frequency. Figure 8 shows measured vertical displacement of the wheel and vehicle body over the frequency range. The damped primary ride mode appears as a dominant resonant peak near 1 Hz on the body displacement curve, and the damped wheel hop mode appears as a dominant resonant peak near 9.5 Hz on the wheel displacement curve. These observed resonant frequencies are consistent with literature [1] as well as a calculation of natural frequencies using a simple quarter-car model with effective wheel rates and wheel and vehicle body masses [1] using parameters in Table 1.

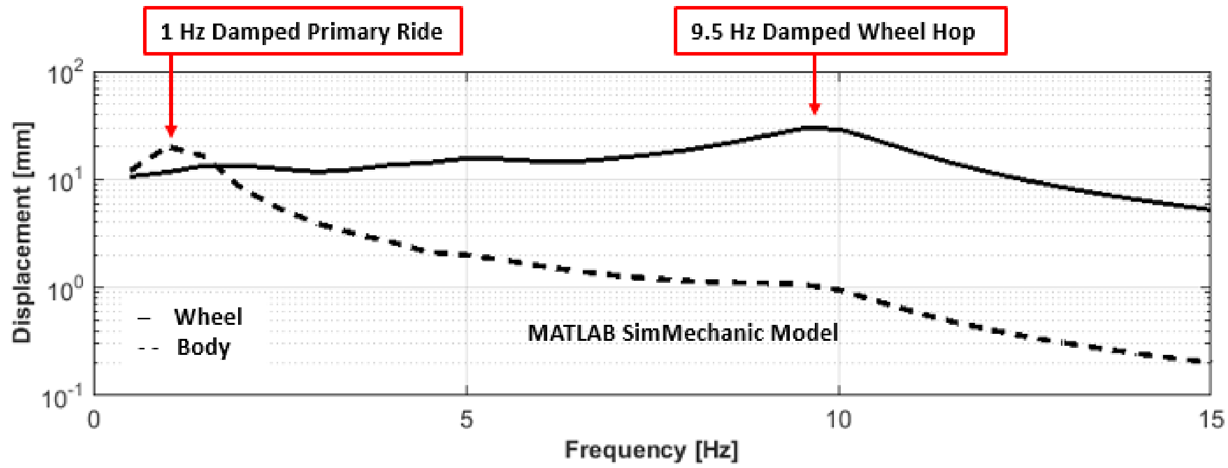

Figure 8. Vertical Displacement vs. Frequency for Wheel and Vehicle Body

2.4. Development of Virtual Test Profiles

In order to investigate the vehicle body stiffness modification on the vehicle ride comfort and handling performance, virtual test profiles are designed with defined lateral and vertical inputs to simulate the respective phenomena. A test profile consists of a controlled harmonic displacement input applied at the four actuators with translational accelerations and angular velocities measurements at the center mass of the vehicle. The peak-to-peak values of measured accelerations and angular velocities are reported for each test in the steady state portion of the response curves, well after the model start-up transients have dissipated. The different virtual test profiles are summarized in Figure 9.

To evaluate vehicle handling performance, a lateral input is selected to simulate a slalom lateral vehicle maneuver at 35 mph. The input is a constant amplitude sine wave with displacement Δ_y equivalent to 0.6 G acceleration at a 3 Hz excitation frequency applied at each actuator in the lateral (Y) direction. The rear wheel actuators lag the front wheel actuators by $\tau = 0.2$ seconds. The 3 Hz frequency is selected since it is at the onset of the observed yaw resonance of the model (from a stepped-sine frequency sweep with a 0.03 G constant acceleration lateral input at the actuators, shown in Figure 10). The peak-to-peak lateral acceleration a_{ypp} (in m/s^2), yaw angular velocity ω_{rzpp} (in rad/s), and roll angular velocity ω_{rxpp} (in rad/s) is measured at the center mass of the vehicle. In addition, the wheel displacements and forces are checked to ensure that realistic vehicle loads are achieved and that there is no loss of contact from the wheels to the actuators at the tire contact patches.

To evaluate vehicle ride performance, a vertical input is selected to simulate a car driving on a wavy road, which also induces a twisting of the vehicle body. The input is a constant 25 mm displacement Δ_z amplitude sine wave applied at each actuator at 3 Hz frequency in the vertical (Z) direction. The inputs for the front left and rear right actuators are 180 degrees out-of-phase with the input for the rear left and front right actuators, effectively maximizing the twisting of the vehicle. The peak-to-peak vertical acceleration a_{zpp} (in m/s^2), pitch angular velocity ω_{rypp} (in rad/s), and roll angular velocity ω_{rxpp} (in rad/s) is measured at the center mass of the vehicle. In addition, the wheel displacements and forces are checked to ensure that realistic vehicle loads are achieved and that there is no loss of contact from the wheels to the actuators at the tire contact patches.

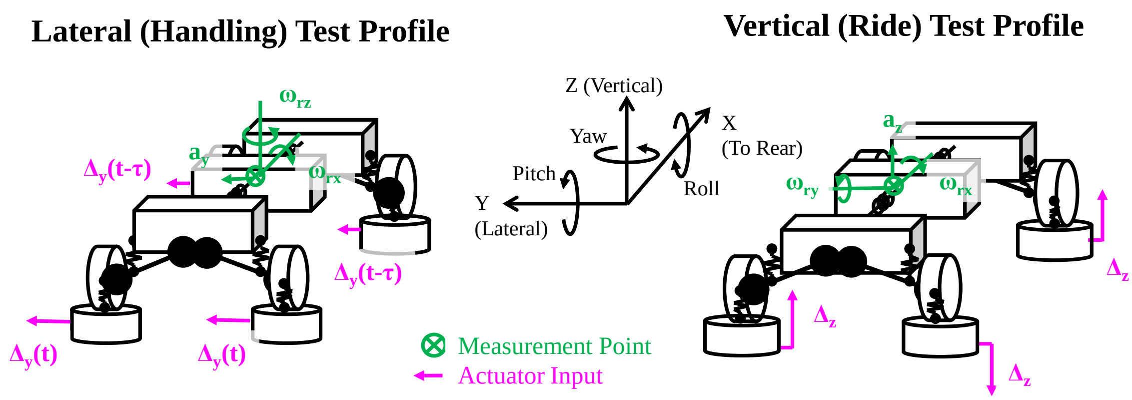

Figure 9. Illustration of Virtual Vehicle Test Profiles

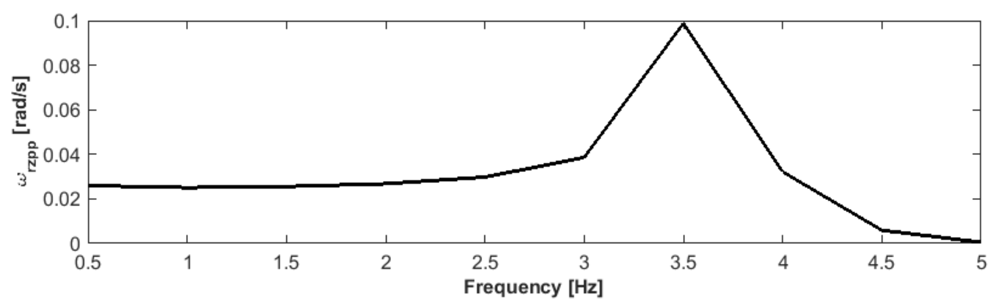

Figure 10. Vehicle Yaw Rotational Velocity from a 0.03 G Lateral Stepped-Sine Frequency Sweep

3. Results

In this section, the effect of modification of vehicle body stiffness on vehicle ride and handling performance is evaluated by systematically changing the vehicle body stiffness and recording the corresponding output responses. Front vehicle body stiffness, rear body stiffness, and the combination of front and rear vehicle body stiffness are evaluated separately for both lateral and vertical test cases. In addition, a sensitivity study is conducted to compare the influence of body stiffness modification to realistic variations in vehicle parameters, such as compact patch stiffness and damper performance curves.

3.1. Lateral Test Results

Figure 11 shows the effect of modifying front vehicle body stiffness on the vehicle dynamic performance for the lateral test profile. Here, the rear body stiffness is kept at the nominal value while the front body stiffness is modified. As the results show, for the same actuator inputs, the vehicle lateral acceleration, yaw angular velocity, and roll angular velocity increase as front vehicle body stiffness decreases. For the same actuator inputs, the vehicle lateral acceleration, yaw angular velocity, and roll angular velocity slightly decreases as front vehicle body stiffness increases. For a more responsive vehicle (improved handling), higher lateral acceleration and yaw angular velocity for the same actuator input are desired. A higher roll angular velocity is also usually desired as it indicates a more responsive force transfer among different corners of the vehicle. As shown in the figure, decreasing the body stiffness too much negatively influences the vehicle handling performance. Therefore, in order to improve vehicle handling performance, the front vehicle body stiffness should be decreased by no more than 50% from the nominal case.

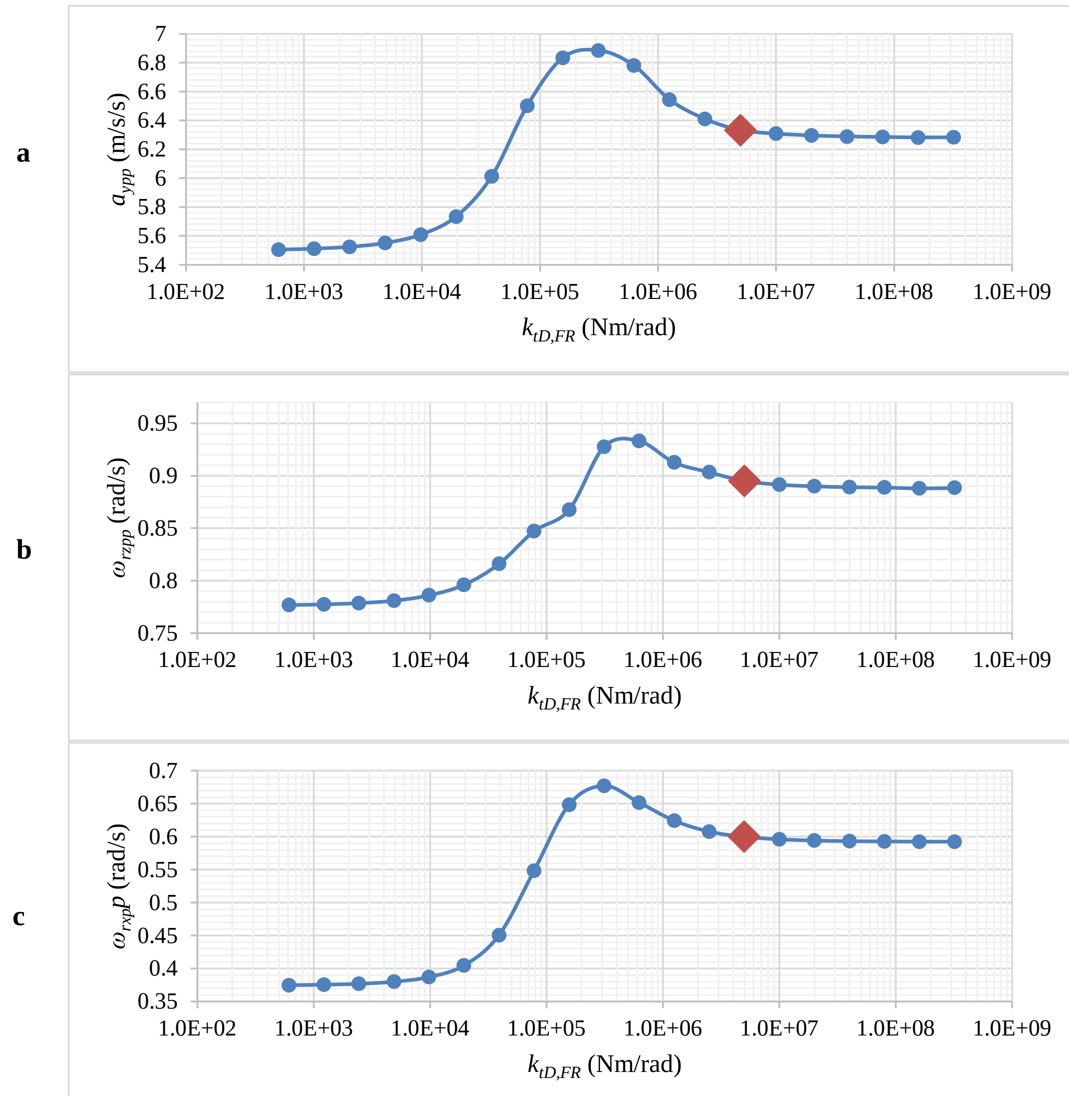

Figure 11. (a) Lateral Acceleration, (b) Yaw Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Front Vehicle Body Stiffness in the Lateral Test; ♦ Indicates the Nominal Case

Figure 12 shows the effect of modifying rear vehicle body stiffness on the vehicle dynamic performance for the lateral test profile. Here, the front body stiffness is kept at the nominal value while the rear body stiffness is modified. As the results show, the vehicle lateral acceleration and yaw angular velocity increase as the rear body stiffness decreases. As shown in the figure, decreasing the body stiffness too much negatively influences the vehicle handling performance. Therefore, in order to improve vehicle handling performance, the rear vehicle body stiffness should be decreased by no more than 50% from the nominal case. In this case, this reduction in vehicle body stiffness has a minimal effect on roll angular velocity.

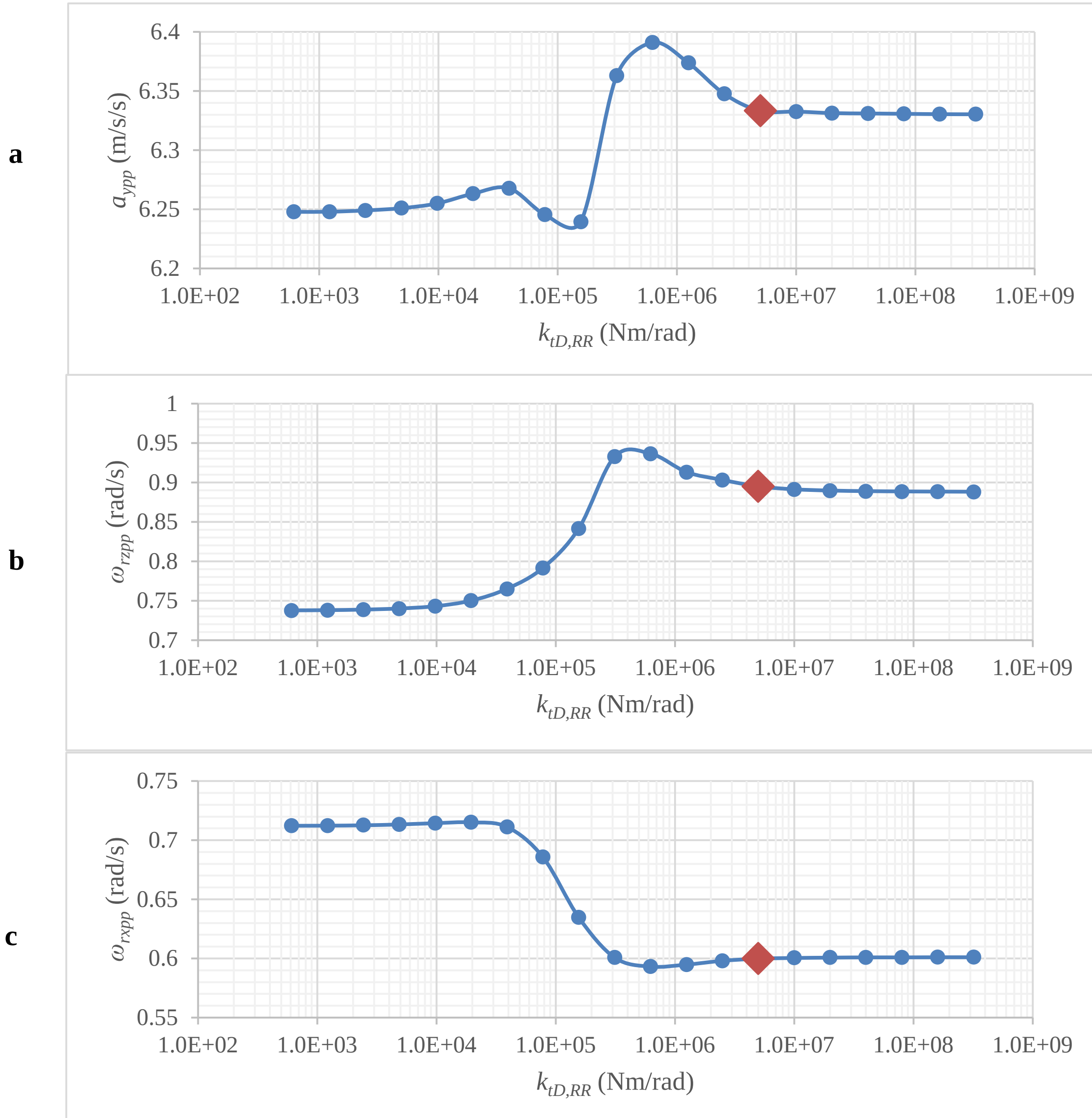

Figure 12. (a) Lateral Acceleration, (b) Yaw Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Rear Vehicle Body Stiffness in the Lateral Test; ♦ Indicates the Nominal Case

Figure 13 shows the effect of simultaneously modifying both the front and rear vehicle body stiffness parameters on the vehicle dynamics performance for the lateral test profile. As the results show, the vehicle lateral acceleration, yaw angular velocity, and roll angular velocity generally increases as both the front and rear vehicle body stiffness parameters decrease. Again, as shown in the figure, decreasing the body stiffness too much negatively influences the vehicle handling performance. Therefore, in order to improve vehicle handling performance, the vehicle body stiffness should be decreased by no more than 50% from the nominal case. In this case, the effect of vehicle body stiffness on vehicle performance more closely resembles the behavior of a front vehicle body modification, as shown in Figure 11, rather than the behavior of a rear vehicle body modification, as shown in Figure 12.

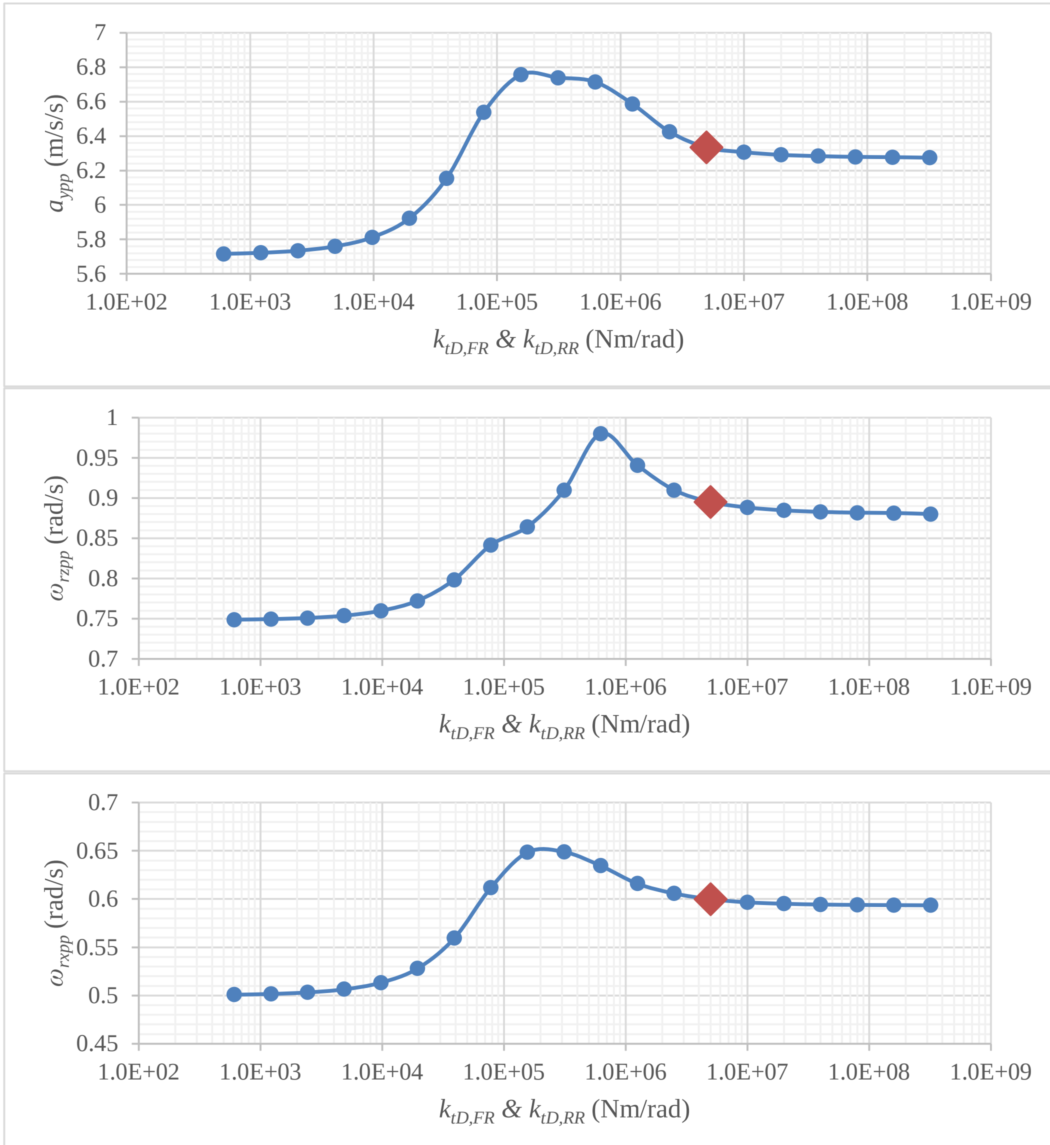

Figure 13. (a) Lateral Acceleration, (b) Yaw Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Vehicle Body Stiffness in the Lateral Test; ♦ Indicates the Nominal Case

3.2. Vertical Test Results

Figure 14 shows the effect of modifying front vehicle body stiffness on the vehicle dynamic performance for the vertical test profile. Here, the rear body stiffness is kept at the nominal value while the front body stiffness is modified. As the results show, for the same actuator inputs, the vehicle vertical acceleration, pitch angular velocity, and roll angular velocity decreases as front vehicle body stiffness decreases. For the same actuator inputs, the vehicle vertical acceleration, pitch angular velocity, and roll angular velocity slightly increases as front vehicle body stiffness increases. For a softer ride (improved ride comfort), lower vertical acceleration, pitch angular velocity, and roll angular velocity for the same actuator input are desired. As shown in the figure, decreasing the body stiffness too much negatively influences the vehicle ride comfort performance. Therefore, in order to improve vehicle ride comfort performance, the front vehicle body stiffness should decreased by no more than 50% from the nominal case.

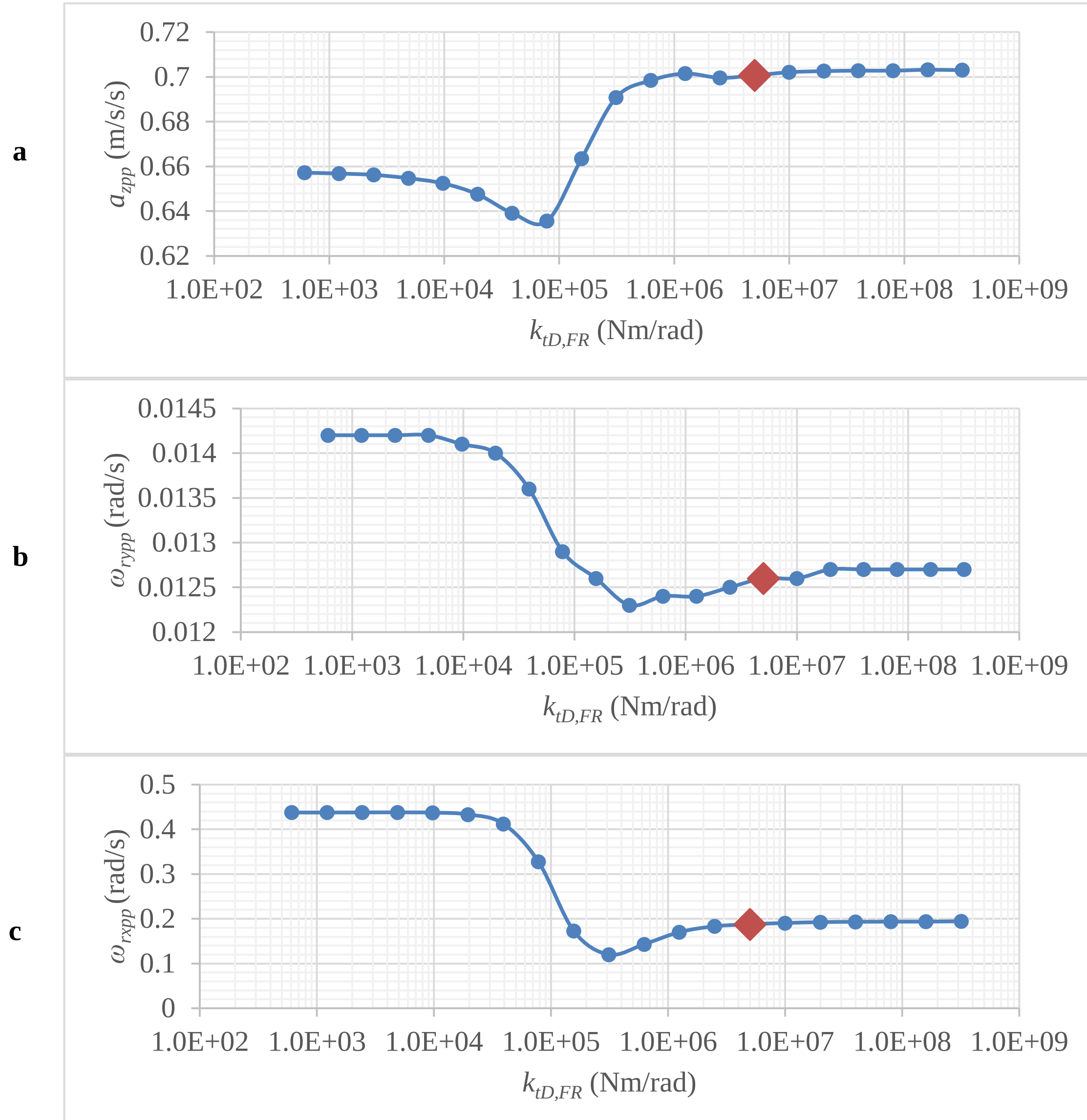

Figure 14. (a) Vertical Acceleration, (b) Pitch Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Front Vehicle Body Stiffness in the Vertical Test; ♦ Indicates the Nominal Case

Figure 15 shows the effect of modifying rear vehicle body stiffness on the vehicle dynamic performance for the vertical test profile. Here, the front body stiffness is kept at the nominal value while the rear body stiffness is modified. As the results show, the vehicle vertical acceleration decreases as the rear body stiffness decreases. However, the pitch angular velocity and roll angular velocity increase as rear vehicle body stiffness decreases. As shown in the figure, decreasing the body stiffness too much negatively influences the vehicle ride comfort performance, with respect to vertical acceleration.

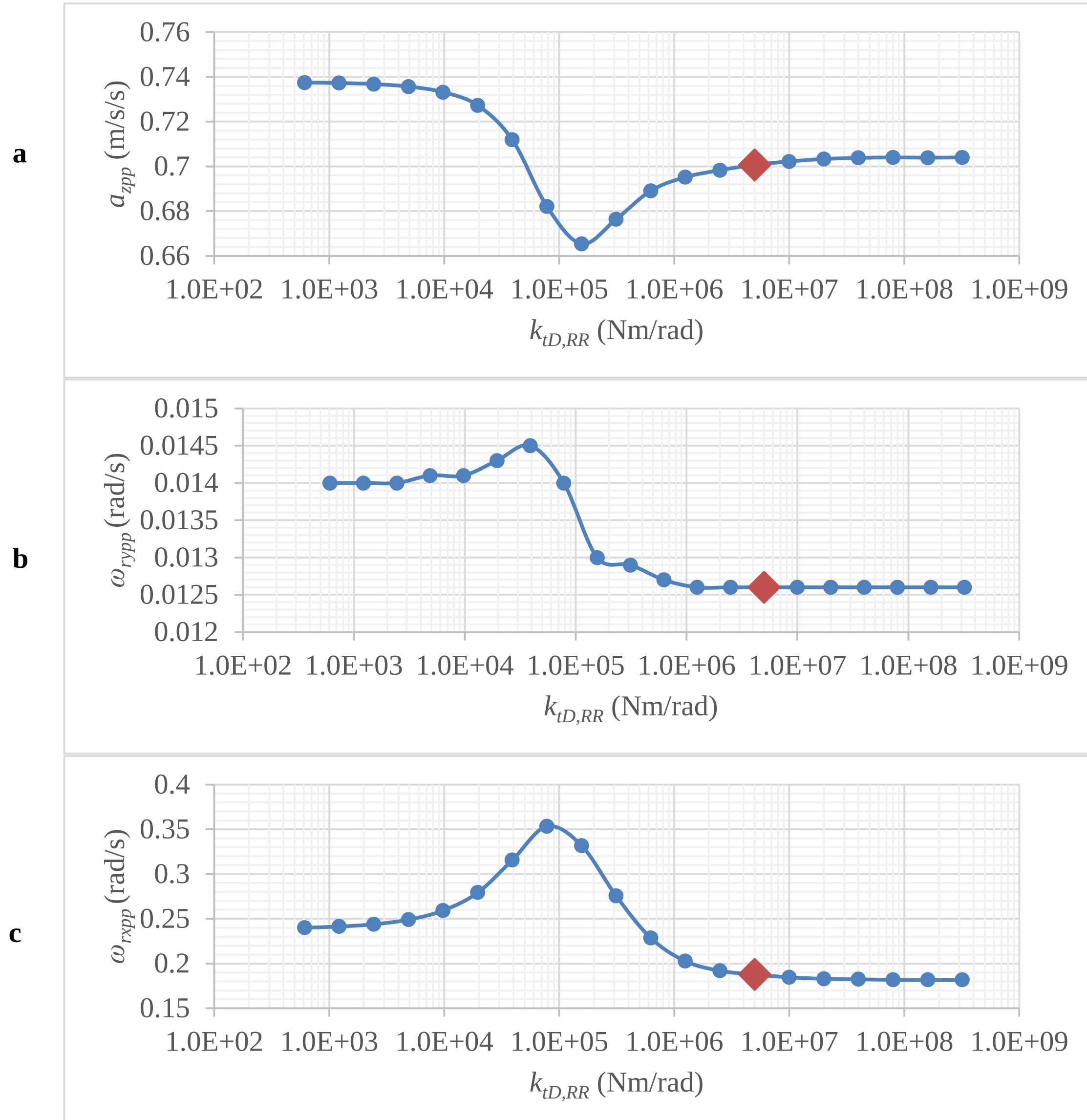

Figure 15. (a) Vertical Acceleration, (b) Pitch Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Rear Vehicle Body Stiffness in the Vertical Test; ♦ Indicates the Nominal Case

Figure 16 shows the effect of simultaneously modifying both the front and rear vehicle body stiffness parameters on the vehicle dynamics performance for the lateral test profile. As the results show, the vehicle vertical acceleration and roll angular velocity decrease when both the front and rear vehicle body stiffness decrease. Therefore, in order to improve vehicle ride comfort performance, the vehicle body stiffness should be decreased by no more than 50% from the nominal case. In this case, this reduction in vehicle body stiffness has a minimal effect on pitch angular velocity.

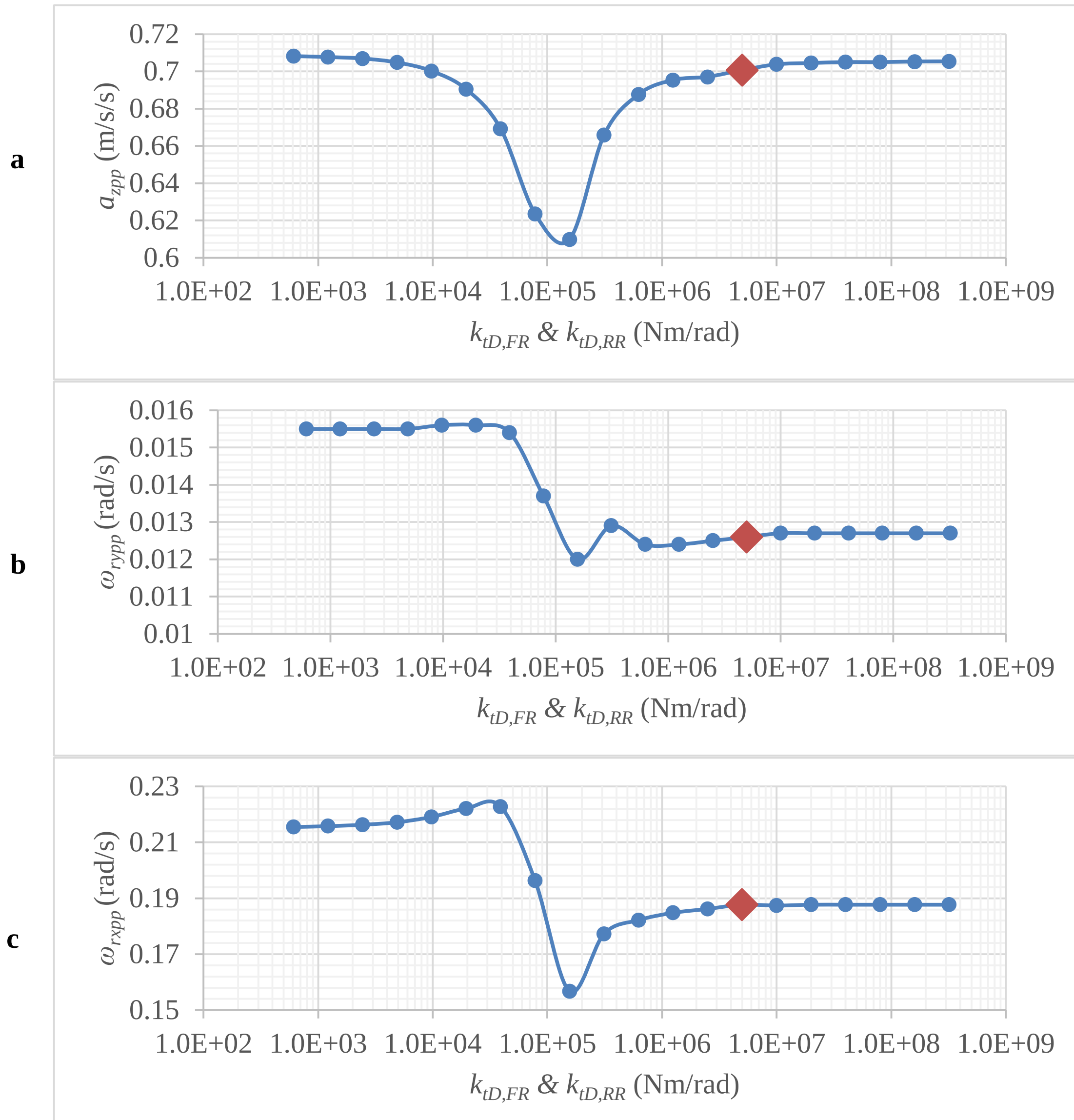

Figure 16. (a) Vertical Acceleration, (b) Pitch Angular Velocity, and (c) Roll Angular Velocity of the Vehicle Middle Body vs. Vehicle Body Stiffness in the Vertical Test; ♦ Indicates the Nominal Case

3.3. Sensitivity Study

Next, a sensitivity study is conducted to compare the influence of body stiffness modification to realistic variations in vehicle parameters, such as compact patch stiffness and damper performance curves. Often these parameters can vary within 25% (as a conservative estimate) of a specified performance, considering variations due to temperature, wear, or manufacturing/assembly. These parameters can also change during the development process as a physical vehicle prototype is tuned by modifying or evaluating different components to achieve a desired vehicle performance. The sensitivity of the parameters in the model are quantified by independently increasing or decreasing each parameter by 25% from the nominal value used in the model. Then, an absolute change of the resulting vehicle dynamic performance for each parameter is calculated relative to the nominal case.

Figure 17 and Figure 18 show the results of sensitivity study for the lateral test cases. By changing tire contact patch stiffness (k.CP) parameters by 25 %, the maximum change in lateral acceleration output is 1.41 m/s^2. By changing suspension damping (fd.BC) by 25 %, the maximum change in lateral acceleration output is 0.42 m/s^2. However, changing vehicle body stiffness (kt.D) parameters by 25 %, the lateral acceleration output is changed by a maximum of 0.024 m/s^2, over an order of magnitude less than the tire contact patch and suspension damping. Therefore, vehicle body stiffness parameters are insensitive parameters when compared with realistic variations in the contact patch stiffness and suspension damping parameters.

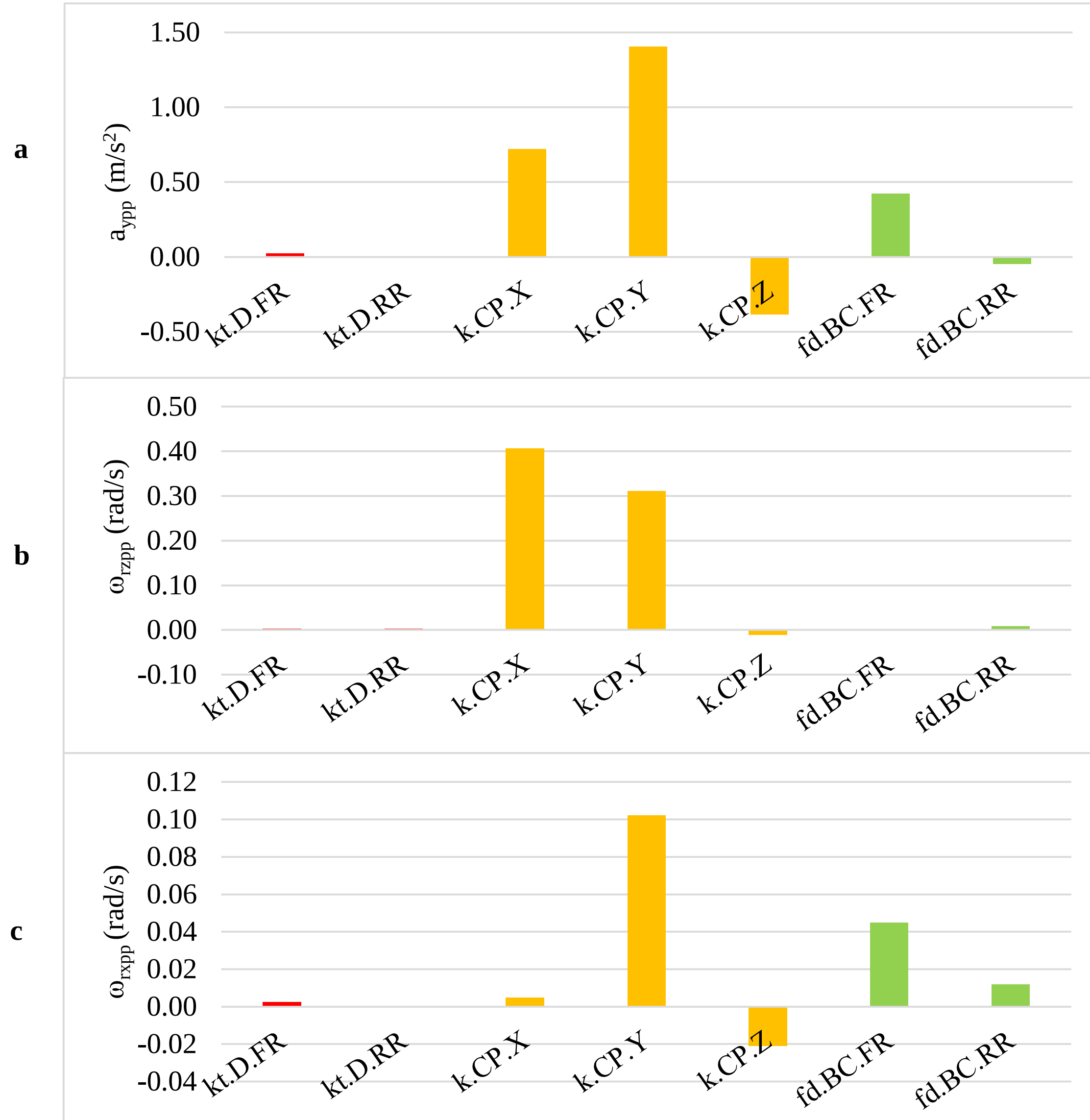

Figure 17. Change of (a) Lateral Acceleration, (b) Yaw Angular Velocity, and (c) Roll Angular Velocity of Middle Vehicle Body for a 25% Decrease in Parameters in the Lateral Test

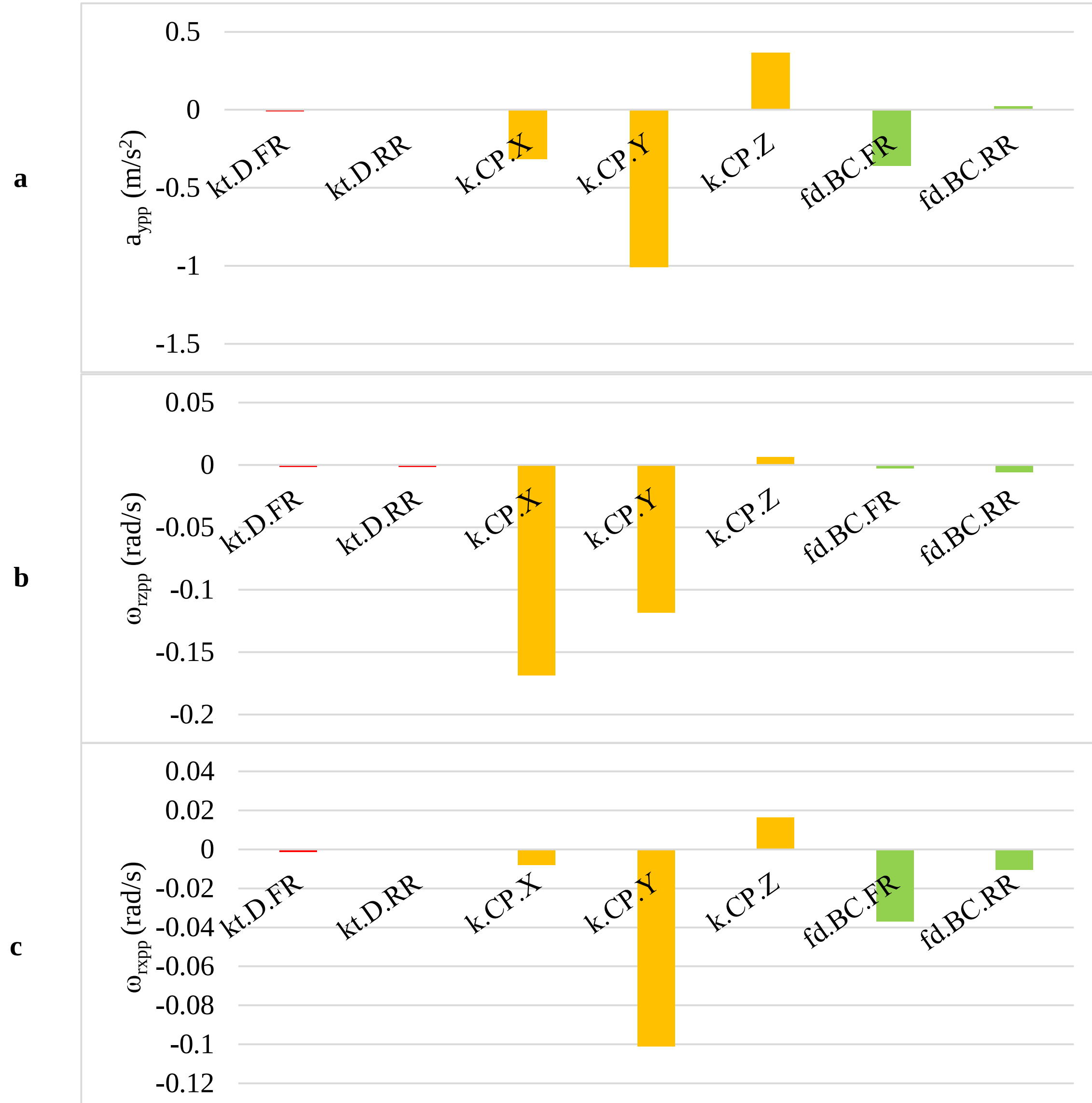

Figure 18. Change of (a) Lateral Acceleration, (b) Yaw Angular Velocity, and (c) Roll Angular Velocity of Middle Vehicle Body for a 25% Increase in Parameters in the Lateral Test

Figure 19 and Figure 20 show the results of sensitivity study for the vertical test cases. By changing tire contact patch stiffness (k.CP) parameters by 25 %, the maximum change in vertical acceleration output is 0.18 m/s^2. By changing suspension damping (fd.BC) by 25 %, the maximum change in vertical acceleration output 0.061m/s^2. However, changing vehicle body stiffness (kt.D) parameters by 25 %, the vertical acceleration output is changed by a maximum of 0.0005 m/s^2, over an order of magnitude less than the contact patch and suspension damping. Again, vehicle body stiffness parameters are insensitive parameters when compared with realistic variations in the tire contact patch stiffness and suspension damping parameters.

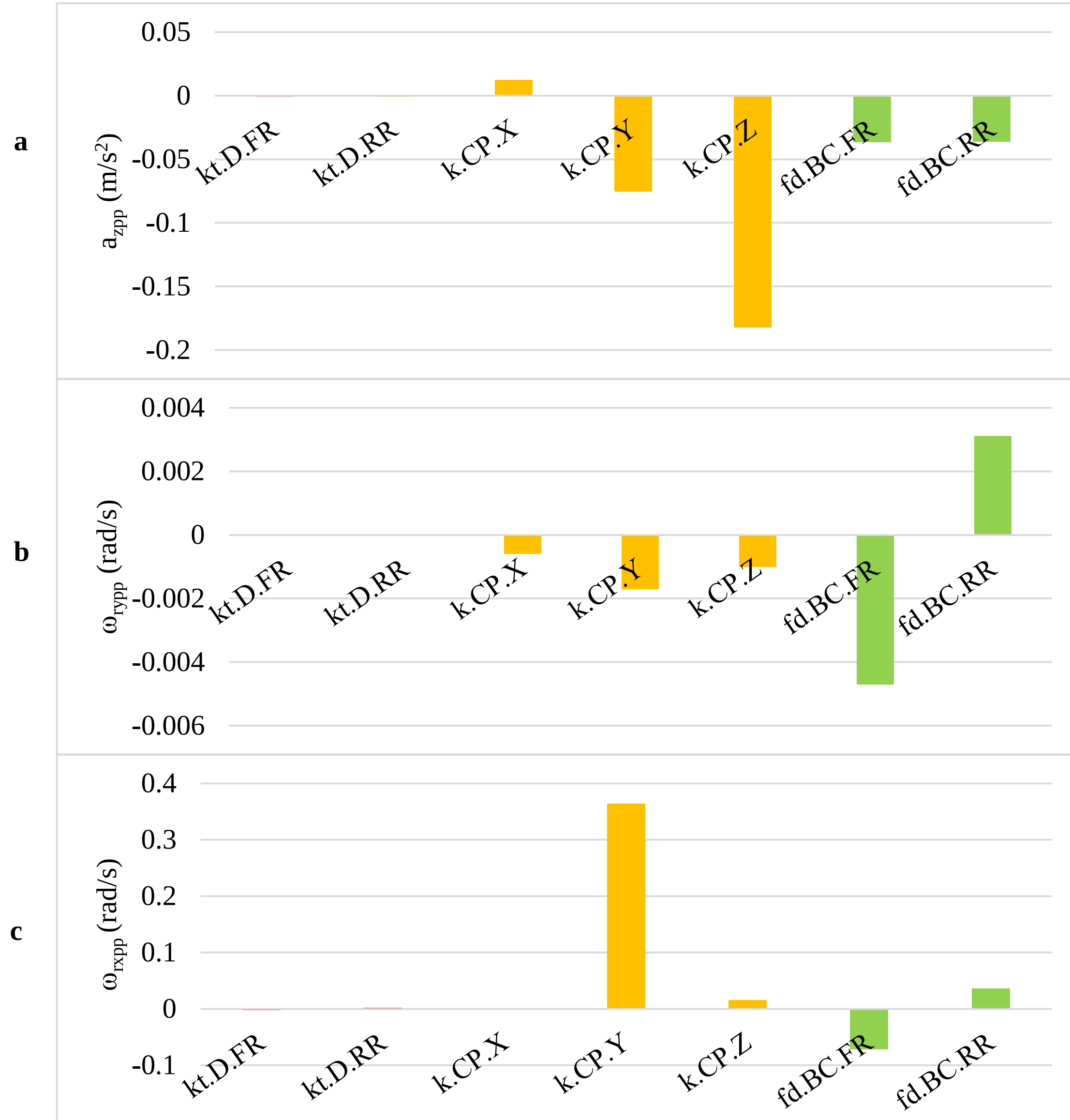

Figure 19. Change of (a) Vertical Acceleration, (b) Pitch Angular Velocity, and (c) Roll Angular Velocity of Middle Vehicle Body for a 25% Decrease in Parameters in the Vertical Test

a

b

c

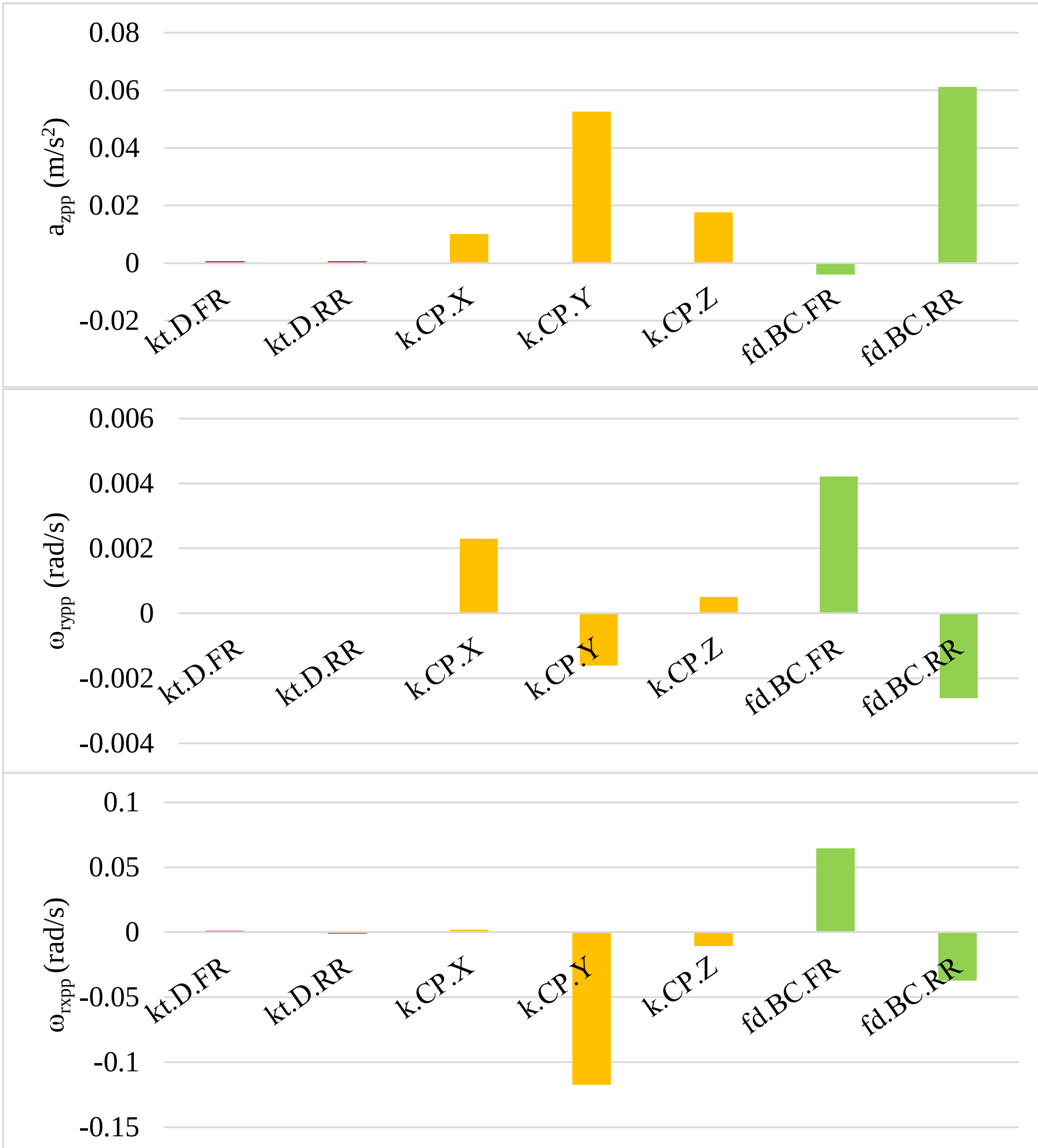

Figure 20. Change of (a) Vertical Acceleration, (b) Pitch Angular Velocity, and (c) Roll Angular Velocity of Middle Vehicle Body for a 25% Increase in Parameters in the Vertical Test

4. Conclusion

4.1. Summary

In this thesis, a reduced-order MBD vehicle model is developed to include vehicle body stiffness representations. The parameters are adapted from an existing MBD model and modal data reported in literature [1, 8], and the model is successfully implemented using MATLAB SimMechanics along with a virtual test procedure to assess vehicle ride comfort and handing performance. The effects of the vehicle body stiffness distribution is investigated by systematically varying model parameters and quantifying the dynamic response of the vehicle body at its center of mass.

4.2. Major Conclusions

For the vehicle considered, improvements to both ride and handling could be achieved through decreasing vehicle body stiffness by upwards of 50%; however, in comparison to realistic variations in tire contact patch and shock absorber parameters, modifications to the body stiffness are minimal. In addition, variation of both the front vehicle body stiffness seems to provide the most consistent improvement for both ride comfort and handling performance. Although changes in vehicle body stiffness parameters are found to be insensitive as part of this study, the tractable modeling approach from this research could be used in low-order vehicle design tools to quickly assess the influence of vehicle body stiffness on the ride comfort and handling performance of future vehicle designs.

4.3. Future Work

This model is limited by the many simplifying assumptions, such as neglecting additional kinematics and compliance behavior by using the simplified swing arm suspension, modeling the tire as a Cartesian joint at the contact patch, and simplifying the stiffness distribution of the vehicle body by constraining the flexural motion to rotation about a single axis. Overcoming these assumptions will require increasing the model fidelity (DOFs), which is outside of the scope of this current study. In addition, this model can not describe how to physically realize vehicle body stiffness modifications; however, it does provide some insight as to which region of the vehicle to focus the structural modification efforts.

Finally, future work required to better understand the effect of vehicle body stiffness modification on vehicle dynamic performance could include the following:

- Study of a vehicle with a response that is more sensitive to body stiffness modifications;
- Inclusion of the powertrain bodies in the model and evaluation their effects on the vehicle performance;
- Model validation on a physical vehicle in a similar fashion as previous literature using strain gages or structural modifications [2, 3];

- Addition of fidelity to the model (DOFs) and investigation of the interaction with suspension parameters, such as layout / location of hard points, joint stiffness and damping, or rolling tire models; and
- Evaluation of the effect of preload of the suspension on the vehicle response (i.e. changes to operating suspension kinematics and vehicle center of mass).

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